Notice No.4

Rules and Regulations for the Classification of Special Service Craft

July 2020

The status of this Rule set is amended as shown and is now to be read in conjunction with this and prior Notices. Any corrigenda included in the Notice are effective immediately.

Please note that corrigenda amends to paragraphs, Tables and Figures are not shown in their entirety.

Issue date: June 2021

Amendments to	**Effective date**	**IACS/IMO implementation (if applicable)**
Part 10, Chapter 1, Section 3	Corrigenda	N/A
Part 16, Chapter 1, Sections 1 & 2	Corrigenda	N/A

Lloyd's Register

Part 10, Chapter 1
Reciprocating Internal Combustion Engines

■ *Section 3*
Crankshaft Design

3.2 Scope

3.2.9 Further information and guidance on crankshaft design is provided in the LR~~'s~~ *Guidance Notes for the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests* *for Crankshafts*.

3.3 Information to be submitted

3.3.2 The following information is also required for appraisal of the crankshaft (not contained in Form 2073):

- every surface treatment affecting fillets or oil holes shall be specified so as to enable calculation according to Chapter ~~2~~ 3 of the LR ~~*Guidance Notes for Crankshaft SCF Calculation using Finite Element Method*~~ Guidance Notes *for the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts*;
    - o this is to include ~~Crankshaft~~ crankshaft fatigue enhancement factors ~~K_1~~ K_1 and ~~K_2~~ K_2 where applicable.

3.4 Symbols

3.4.1 For the purposes of this Chapter the following symbols apply, *see also*;

D_A = the outside diameter of we or twice the minimum distance between ~~centre line~~ centreline of journals and outer contour of web, whichever is less, in mm

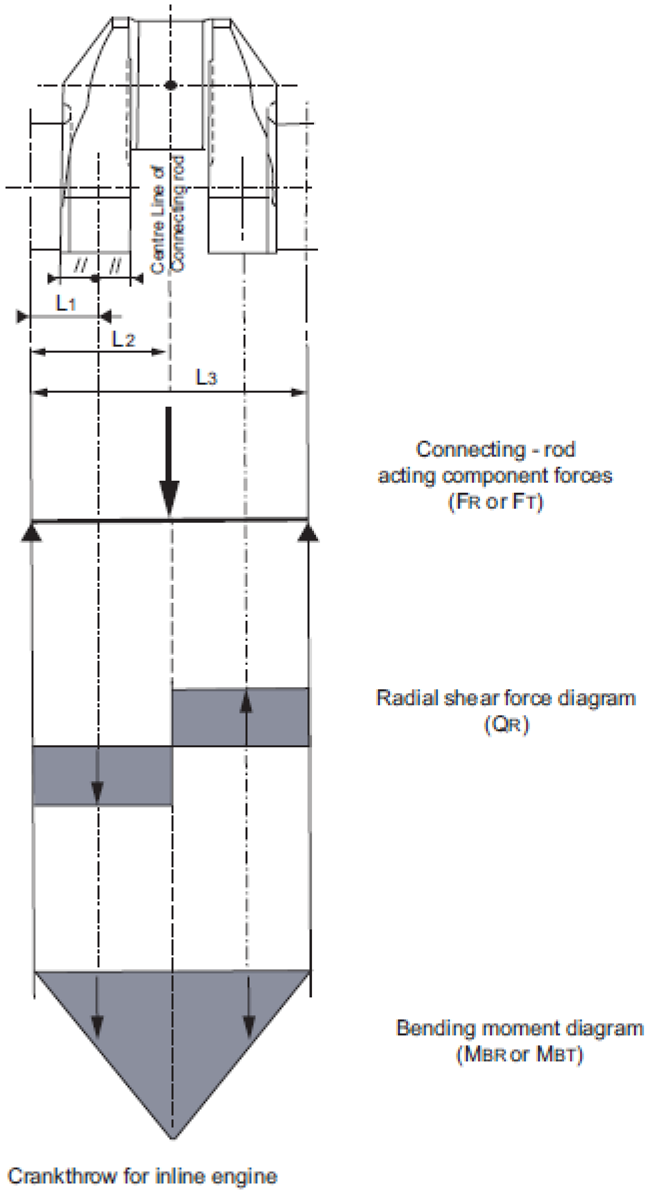
y = distance between the adjacent generating lines of journal and pin, in mm

Note for $y \geq 0{,}5D_s$ ~~where,~~. Where y is less than $0{,}1D_s$, special consideration is to be given to the effect of the stress due to the shrink fit on the fatigue strength at the crankpin fillet.

σ_y = equivalent alternating stress for crankpin fillet, journal fillet or outlet of crankpin oil bore as applicable, in ~~N/mm^3~~ N/mm^2

3.5 Calculation of alternating stresses due to bending moments and radial forces – assumptions

Figure 1.3.9 Bending moment and shear force for in-line engine crankthrows has been replaced with the below figure;



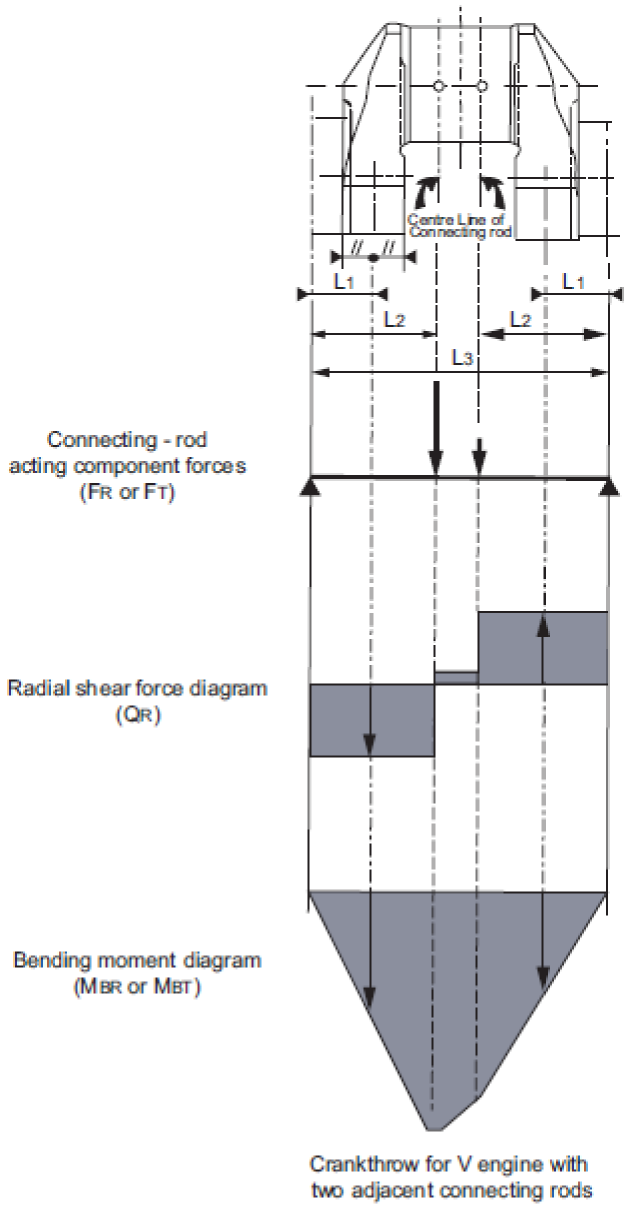
L_1 = Distance between main journal centreline and crankweb centre

L_2 = Distance between main journal centreline and connecting - rod centre

L_3 = Distance between two adjacent main journal centrelines

Figure 1.3.9 Bending moment and shear force for in-line engine crankthrows

Figure 1.3.10 Bending moment and shear force for V engine crankthrows has been replaced with the below figure;


Centre Line of Connecting rod
L1
L2
L3
L1
L2
Connecting - rod acting component forces (FR or FT)
Radial shear force diagram (QR)
Bending moment diagram (MBR or MBT)
Crankthrow for V engine with two adjacent connecting rods

L_1 = Distance between main journal centreline and crankweb centre

L_2 = Distance between main journal centreline and connecting - rod centre

L_3 = Distance between two adjacent main journal centrelines

Figure 1.3.10 Bending moment and shear force for V engine crankthrows

3.5.4 ~~The two relevant bending moments for bending acting on the outlet of crankpin oil bores are taken in the crankpin cross-section through the oil bore.~~ *~~See~~* ~~*Figure 1.3.9 Bending moment and shear force for in-line engine crankthrows* and *Figure 1.3.10 Bending moment and shear force for V engine crankthrows*.~~ ~~M_{BRO} is the bending moment of the radial component of the connecting-rod force and M_{BTO} is the bending moment of the tangential component of the connecting-rod force. The alternating stresses due to these bending moments are to be related to the cross-sectional area of the axially bored crankpin. Mean bending stresses are neglected~~

The two relevant bending moments for bending acting on the outlet of crankpin oil bores are taken in the crankpin cross-section through the oil bore. *See;*

- *Figure 1.3.5 Crankpin section through the oil bore*,
- *Figure 1.3.9 Bending moment and shear force for in-line engine crankthrows*, and
- *Figure 1.3.10 Bending moment and shear force for V engine crankthrows*.

M_{BRO} is the bending moment of the radial component of the connecting-rod force and M_{BTO} is the bending moment of the tangential component of the connecting-rod force. The alternating stresses due to these bending moments are to be related to the cross-sectional area of the axially bored crankpin. Mean bending stresses are neglected.

3.6 Calculation of bending stresses

3.6.1 The radial and tangential forces due to gas and inertia loads acting upon the crankpin at each connecting-rod position are to be calculated over one working cycle. Using the forces calculated over one working cycle and taking into account of the distance from the main bearing midpoint, the time curve of the bending moments, M_{BRF}, M_{BRO} and M_{BTO}, and radial forces, Q_{RF}, as defined in *Pt 10, Ch 1,* *3.5 Calculation of alternating stresses due to bending moments and radial forces – assumptions ~~3.5.2~~ 3.5.3* and *Pt 10, Ch 1, 3.5 Calculation of alternating stresses due to bending moments and radial forces – assumptions ~~3.5.3~~ 3.5.4* are then calculated.

3.6.6 Nominal alternating bending and compressive stresses in a web cross-section are calculated as follows:

$$\sigma_{BFN} = \pm\frac{M_{BRFN}}{W_{eqw}} 10^3 \; K_e \text{ N/mm}^2$$

$$\sigma_{QFN} = \pm\frac{Q_{RFN}}{F} \; K_e \text{ N/mm}^2$$

where

~~$M_{BRFN} = \pm\frac{1}{2}(X_{BRF\,max} - X_{BRF\,min})\text{Nm}$~~ $M_{BRFN} = \pm\frac{1}{2}(M_{BRF\,max} - M_{BRF\,min})\text{Nm}$

~~$Q_{RFN} = \pm\frac{1}{2}(Q_{RF\,max} - Q_{RF\,min})\text{Nm}$~~ $Q_{RFN} = \pm\frac{1}{2}(Q_{RF\,max} - Q_{RF\,min})\text{N}$

3.6.7 Nominal alternating bending stress in the outlet of the crankpin oil bore is calculated as follows:

$$\sigma_{BON} = \pm\frac{M_{BON}}{W_e} 10^3 \text{ N/mm}^2$$

where

M_{BON} is taken as the ½ range value M_{BON} = ± ½ (M_{BOmax} – M_{BOMin}) Nm

3.7 Calculation of torsional stresses

3.7.1 The nominal alternating torsional stress, ~~τ_n~~ τ_N, is to be taken into consideration. The value is to be derived from forced-damped vibration calculations of the complete dynamic system. Alternative methods will be given consideration. The engine designer is to advise the maximum level of alternating vibratory stress that is permitted (τ_a).

3.7.2 τ_a or ~~τ_n~~ τ_N(as applicable) is to be applied as a limiting value for the torsional vibration assessment required by *Pt 13, Ch 1, Torsional vibration*.

3.7.3 Nominal alternating torsional stress is calculated as follows:

~~τ_n~~ $\tau_N = \frac{M_{TN}}{W_P} 10^3$ N/mm^2

where

~~$W_p = \frac{\pi}{16}\left(\frac{D^4 - D_{BH}{}^4}{D}\right)$ mm^3 for the crankpin, or $W_p = \frac{\pi}{16}\left(\frac{D_G{}^4 - D_{BG}{}^4}{D_c}\right)$ mm^3 for the jorunal~~

$W_p = \frac{\pi}{16}\left(\frac{D^4 - D_{BH}{}^4}{D}\right)$ mm^3 for the crankpin, or $W_p = \frac{\pi}{16}\left(\frac{D_G{}^4 - D_{BG}{}^4}{D_c}\right)$ mm^3 for the journal

3.8 Stress concentration factors

3.8.3 Where the geometry of the crankshaft is outside the boundaries (*see Table 2.3.2 Crankshaft variable boundaries for analytical SCF calculation*) of the analytical SCF the calculation method detailed in Chapter 1 and Chapter 4 of the LR *Guidance Notes for ~~Crankshaft SCG Calculation using Finite Element Method~~ Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts* may be undertaken.

3.8.5 Chapters1 and ~~3~~ 4 of the LR *Guidance Notes for ~~Crankshaft SCG Calculation using Finite Element Method~~ Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts* describe how finite element (FE) analyses can be used for the calculation of the SCF. Care needs to be taken to avoid mixing equivalent (von Mises) stresses and principal stresses.

3.8.6 Crankpin SCF are calculated as follows:

a) Bending

~~$\alpha_B = 2{,}6914 f(s,w).f(w).f(b).f(r).f(d_G).f(d_H).f(\text{recess})$~~

$\alpha_B = 2{,}6914 \cdot f(s,w) \cdot f(w) \cdot f(b) \cdot f(r) \cdot f(d_G) \cdot f(d_H) \cdot f(\text{recess})$

b) Torsion

~~$\alpha_T = 0{,}8 f(r,s).f(b).f(w)$~~

$\alpha_T = 0{,}8 \cdot f(r,s) \cdot f(b) \cdot f(w)$

where

$f(b) = 7{,}8955 - 10{,}654b + 5{,}3482b^2 – 0{,}857b^3$ ~~$f(w) = w^{(-0{,}145)}$~~

$f(w) = w^{(-0{,}145)}$

3.8.7 Journal fillet SCF are calculated as follows (not applicable to semi-built) crankshafts):

a) Bending

~~$\beta_B = 2{,}7146 f_B(s,w).f_B(w).f_B(b).f_B(r).f_B(d_G).f_B(d_H).f(\text{recess})$~~

$\beta_B = 2{,}7146 \cdot f_B(s,w) \cdot f_B(w) \cdot f_B(b) \cdot f_B(r) \cdot f_B(d_G) \cdot f_B(d_H) \cdot f(\text{recess})$

b) Compression due to the radial force~~:~~

~~$\beta_Q = 3{,}0128 f_Q(s).f_Q(w).f_Q(b).f_Q(r).f_Q(d_H).f(\text{recess})$~~

$\beta_Q = 3{,}0128 \cdot f_Q(s) \cdot f_Q(w) \cdot f_Q(b) \cdot f_Q(r) \cdot f_Q(d_H) \cdot f(\text{recess})$

3.9 Additional bending stress

3.9.1 In addition to the alternating bending stresses in fillets (*see Pt 10, Ch 1, 3.6 Calculation of bending stresses 3.6.8*) further bending stresses due to misalignment and bedplate deformation as well as due to axial and bending vibrations are to be considered by applying ~~σ_{add}~~ σ_{add} as given by *Table 1.3.3 Additional bending stresses*.

3.10 Equivalent alternating stress

3.10.4 Equivalent alternating stress, σ_v, is defined as:

(a) For the crankpin fillet:

~~$\sigma_v = \pm\sqrt{(\sigma_{BG} + \sigma_{add})^2 + 3\tau_G{}^2}$ N/mm²~~

$\sigma_v = \pm\sqrt{(\sigma_{BH} + \sigma_{add})^2 + 3\tau_H{}^2}$ N/mm³

(b) For the journal fillet:

~~$\sigma_v = \pm\sqrt{(\sigma_{BH} + \sigma_{add})^2 + 3\tau_H{}^2}$ N/mm³~~

$\sigma_v = \pm\sqrt{(\sigma_{BG} + \sigma_{add})^2 + 3\tau_G{}^2}$ N/mm²

3.11 Fatigue strength

3.11.1 The fatigue strength is to be understood as the value of equivalent alternating stress (~~Von~~ von Mises) which a crankshaft can permanently withstand at the most highly stressed points. The fatigue strength can be evaluated by means of the following formulae:

a) Related to the crankpin diameter:

~~$\sigma_{DW} = \pm K(0{,}42\sigma_B + 39{,}3)\left[0{,}264 + 1{,}073D^{-0{,}2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R_X}}\right]$ N/mm²~~

$\sigma_{DW} = \pm K(0{,}42\sigma_B + 39{,}3)\left[0{,}264 + 1{,}073D^{-0{,}2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R_X}}\right]$ N/mm²

b) Related to the journal diameter:

~~$\sigma_{DW} = \pm K(0{,}42\sigma_B + 39{,}3)\left[0{,}264 + 1{,}073D_G{}^{-0{,}2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R_G}}\right]$ N/mm²~~

$\sigma_{DW} = \pm K(0{,}42\sigma_B + 39{,}3)\left[0{,}264 + 1{,}073D^{-0{,}2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R_X}}\right]$ N/mm²

A value for K_2 will be assigned upon application by the engine designers. Full details of the process, together with the results of full scale fatigue tests will be required to be submitted for consideration. *See* Chapter 2 of the *LR Guidance ~~note - Guidance for the evaluation of Crankshaft Fatigue Tests~~ Notes for the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts*. Alternatively, the following values may be taken (surface hardened zone to include fillet radii):

3.11.3 Fatigue strength calculations or, alternatively, fatigue test results determined by experiment based either on full size crankthrow (or crankshaft), or on specimens taken from a full size crankthrow, may be required to demonstrate acceptability. The experimental procedure for fatigue evaluation of specimens and fatigue strength of crankshaft assessment are to be submitted for approval by LR. The procedure is to include as a minimum: method, type of specimens, number of specimens (or crankthrows), number of tests, survival probability, and confidence number. *See also* Chapter 2 of the LR *Guidance ~~for the Evaluation of Crankshaft Fatigue Tests~~ Notes for the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts.*

3.11.5 Only surface treatment processes approved by LR are permitted. Guidance for calculation of surface treated fillets and oil bore outlets is presented in Chapter ~~2~~ 3 of the LR *Guidance Notes for ~~Crankshaft SCF Calculation using Finite Element Method~~ the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts*.

3.13 Shrink fit of semi-built crankshafts

3.13.5 The actual oversize *Z* of the shrink fit must be within the limits Z_{min} and Z_{max} calculated in accordance *Pt 10, Ch 1, 3.13 Shrink fit of semi-built crankshafts ~~3.13.7~~ 3.13.6* and *Pt 10, Ch 1, 3.13 Shrink fit of semi-built crankshafts ~~3.13.8~~ 3.13.7*. When *Pt 10, Ch 1, 3.13 Shrink fit of semi-built crankshafts 3.13.4* cannot be complied with, then the calculated values of Z_{min} and Z_{max} are not applicable due to multizone-plasticity problems. In such cases Z_{min} and Z_{max} are to be established from FEM calculations.

3.13.6 The minimum require diametral ~~Interference~~ interference is to be taken as the greater of:

3.13.7 The maximum diametral interference is not to be greater than:

~~$Z_{\mathrm{max}} \leq D_{\mathrm{S}} \left(\frac{\varrho_{\mathrm{SW}}}{E_{\mathrm{m}}} + \frac{0{,}8}{1000} \right)$ mm~~ $Z_{\mathrm{max}} \leq D_{\mathrm{S}} \left(\frac{\sigma_{\mathrm{SW}}}{E_{\mathrm{m}}} + \frac{0{,}8}{1000} \right)$ mm

Part 16, Chapter 1
Control Engineering Systems

■ *Section 1*
General requirements

1.2 Documentation required for design review

1.2.6 **Programmable electronic systems.**

In addition to the documentation required by *Pt 16, Ch 1, 1.2 Documentation required for design review 1.2.2*, the following is to be submitted:

(g) Software production plans, including applicable procedures, *see* ~~*Pt 16, Ch 1, 2.14 Programmable electronic systems - Additional requirements for integrated systems 2.14.5*~~ *Pt 16, Ch 1, 2.10 Programmable electronic systems – General requirements 2.10.20*.

■ *Section 2*
Essential features for control, alarm, monitoring and safety systems

2.10 Programmable electronic systems – General requirements

2.10.5 ~~Where programmable electronic equipment shares resources, e.g. a data communication link, with any control, alarm or safety system for essential services or safety critical system, software is to meet the requirements of~~ ~~*Pt 16, Ch 1, 2.13 Programmable electronic systems – Additional requirements for essential services and safety critical systems 2.13.7*~~~~.~~ Programmable electronic equipment is to be certified by a recognised authority as suitable for the environmental conditions in which it is intended to operate, *see also Pt 16, Ch 1, 2.13 Programmable electronic systems - Additional requirements for essential services and safety critical systems 2.13.3*.

2.10.20 Software lifecycle activities, e.g. design, development, supply and maintenance, are to be carried out in accordance with an acceptable quality management system which has lifecycle models suitable to the nature of the software project, considering its size, complexity, safety, risk and integrity. Project specific software quality plans are to be submitted. These are to demonstrate that the provisions of ISO/IEC 90003: *Software engineering – Guidelines for the application of ISO 9001:~~2008~~ 2015 to computer software*, or equivalent, are incorporated. The plans are to define responsibilities for the lifecycle activities, including verification, validation, software module testing, integration with other components or systems and security policies to be applied.

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Published by Lloyd’s Register Group Limited
Registered office (Reg. no. 08126909)
71 Fenchurch Street, London, EC3M 4BS
United Kingdom

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